A STUDY OF EXHAUST GAS TEMPERATURE MEASUREMENT METHODS

by

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Thesis
B38

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Submitted in Partial Fulfillment of the Requirements for the

Degree of

Master of Science in

Aeronautical Engineering

from the

Massachusetts Institute of Technology

1947

Thesis
B38

GRADUATE SCHOOL

Cambridge, Massachusetts,
May 23, 1947.

Professor Joseph S. Newell,
Secretary of the Faculty,
Massachusetts Institute of Technology,
Cambridge, Massachusetts.

Dear Sir:

A thesis entitled "A Study of Exhaust Gas Temperature Measurement Methods" is herewith submitted in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

Cambridge, Massachusetts,
May 23, 1917.

Professor Joseph S. Ames,
Secretary of the Faculty,
Massachusetts Institute of Technology,
Cambridge, Massachusetts.

Dear Sir:

I hereby submit "A Study of Relations Between Temperature Coefficient Ratios" in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering.

Respectfully,

ACKNOWLEDGMENTS

Rogowski, A. R. Massachusetts Institute of Technology.
Leary, W. A. " " " "
Ku, P. M. " " " "
Livengood, J. C. " " " "
Kano, C. " " " "
General Electric Company, Schenectady, New York.

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Edgerton, H. E. Massachusetts Institute of Technology.
Jones, F. D. " " " "
Su, T. H. " " " "
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SUMMARY

An investigation was performed on a CFR, single cylinder, four stroke engine to compare three different methods of measuring the temperature of the exhaust gases. The three methods used were:

1. Computation from indicator card.
2. Direct reading of the G.E. shielded thermocouple placed at the exit of the exhaust port.
3. Computation by use of a heat exchanger.

The tests were carried out at three different compression ratios and through a range of five fuel air ratios.

The indicator card method produced the highest temperatures. The G.E. Thermocouple temperature showed an energy content of the exhaust gases lower than the energy content as computed by the indicator method by approximately 6% of the heating value of the fuel. The heat transfer method gave temperatures lower than the other two and approximately ten percent of the fuel heating value below the indicator method. Good correlation was obtained among the three methods in the temperature trends with the maximum temperature at approximately chemically correct fuel air ratio.

The results of these tests are shown as plots of temperature vs percent of chemically correct fuel air ratio. Final conclusions made are that the G.E. Thermocouple is the most practical within the range of the instrument and that the heat exchanger method can be made more accurate than in this study and can be used over a much greater range than the thermo-

An investigation was performed on a GFM, multiple cylinder, four-stroke engine to determine different methods of measuring the temperature of the exhaust gases. The three methods used were:

1. Deflagration type indicator card.
2. Direct reading of the E.G. embedded thermocouple placed at the exit of the exhaust port.
3. Comparative ir ray of a heat chalkboard.

The tests were carried out at three different compression ratios and through a range of five fuel air ratios.

The indicator card method involved the highest percentage. The E.G. Thermocouple temperatures showed an average of the exhaust gases lower than the readings obtained by the indicator method by 8% of the heating value of the fuel. The heat transfer method gave temperatures lower than the other two and approximately ten percent of the fuel heating value below the indicator method. Good correlation was obtained among the three methods at different compression ratios with the maximum temperatures at approximately corrected fuel air ratio.

The results of these tests are plotted on plate at temperature versus the percent of theoretically required fuel air ratio. Final conclusions made are that the E.G. Thermocouple is the most practical within the range of the instrument and that the heat exchanger method may be made more economical than in this study and can be used over a much greater range than the thermo-

couple. The indicator card method is not considered an actual temperature measuring device.

sample. The thermometer bulb method is not considered as accurate
heat temperatures according device.

INTRODUCTION

The measurement of energy in exhaust gases has never been successfully determined due to the inherent difficulty of measuring temperatures in a hot moving stream. Since an accurate measurement of these temperatures is necessary for engine heat balance and for the design of auxiliary turbine units, it is considered extremely important that a method be devised which will give satisfactory results.

The present investigation is a comparison between two of the most satisfactory methods to date, the General Electric shielded thermocouple (Ref. 1), and High Speed Engine Indicator card, with the heat exchanger method of this thesis.

This work was done in the Sloan Automotive Laboratory at the Massachusetts Institute of Technology in April and May, 1947.

ACKNOWLEDGEMENTS

The investigations of energetic ion collisions given herein were successfully performed due to the important utilities of computing facilities in a fast moving atmosphere. Since no accurate measurements of these interactions is necessary for angles less than for the design of auxiliary heating units, it is considered extremely important that a method be developed which will give satisfactory results.

The present investigation is a continuation involving two of the most fundamental problems in data, the Universal Gravitational Thermodynamic (Ref. 1), and High Speed Digital Instrumentation (Ref. 2). This work was done in the Plasma Physics Laboratory at the Massachusetts Institute of Technology in April and May, 1961.

DESCRIPTION OF APPARATUS

GENERAL

The set-up and arrangement of the apparatus used in this investigation is shown in Figs. 1, 2, and 3.

ENGINE

The engine was a CFR, single cylinder, four stroke engine. It was liquid cooled, had single spark ignition, a variable compression ratio, a bore of 3.25 inches, and a stroke of 4.5 inches. It had single inlet and exhaust valves with no overlap. The engine drove, besides the dynamometer, the oil pressure pump, the ignition breaker points, the tachometer, the dynamometer exciter, and the M.I.T. high speed engine indicator.

INLET SYSTEM

Air supplied to the engine was passed through a sharp-edged orifice (0.515 inch diameter - installed with flange taps according to ASME specifications -- Refs. 2 and 3), a 50 gallon surge tank, a throttle valve, a vaporizing tank, and an intake pipe leading to the engine. Temperature of the inlet mixture was controlled by supplying either low pressure steam or cooling water to the vaporizing tank jacket.

FUEL AND FUEL SYSTEM

The fuel used was one hundred octane gasoline supplied by the laboratory system. The fuel was supplied by an externally driven fuel pump and passed through a bubble separator,

[illegible]

[illegible]

The set-up and arrangement of the apparatus used in this investigation is shown in Figs. 1, 2, and 3.

[illegible]

The engine was a one, six-in cylinder, four stroke engine. It was light weight, had double spark ignition, a variable compression ratio, a bore of 1.25 inches, and a stroke of 4.5 inches. It had electric brake and adjustable valve gear to overlap. The engine stand, besides the dynamometer, the oil pressure pump, the ignition booster motor, the tachometer, the dynamometer scales, and the A.C.F. high speed engine indicator.

[illegible]

Air supplied to the engine was passed through a sharp-edged orifice (0.515) inch diameter - installed with flange taps according to ASME specifications -- refs. 2 and 3), a 50 gallon storage tank, a throttle valve, a manometer tank, and an intake side leading to the engine. Temperature of the inlet air/gas was controlled by supplying either low pressure steam or compressed water to the manometer tank jacket.

[illegible] of Fuel System

The fuel used was one hundred octane gasoline supplied by the laboratory stores. The fuel was supplied by an external fuel pump and passed through a Bendix automatic

a rotometer, and a needle valve directly over the vaporizing tank. This valve was used to pressurize the supply line to avoid vapor lock. Vaporization was obtained by heating the vaporizing tank with steam (Fig. 4).

CYLINDER COOLING SYSTEM

Cooling was provided by circulating water through the water jacket by an externally driven circulating pump. The temperature was maintained constant at boiling point and excessive boiling was prevented by cooling coils.

LUBRICATING SYSTEM

The oil pressure was obtained through an engine-driven pump. The oil circulating pump was externally driven. The oil temperature was satisfactorily controlled by a heat exchanger using low pressure steam or water.

IGNITION SYSTEM

Ignition was provided by a breaker operating at one-half crankshaft speed. The electrical circuit is shown in Fig. 5.

EXHAUST SYSTEM

The exhaust gases were discharged from the engine into a 1-1/4 inch iron pipe expanding into a 2 inch pipe in which the General Electric shielded thermocouple was housed (Fig. 6). From here the gases were passed through a flexible steel coupling into a heat exchanger for cooling purposes. The gases from the heat exchanger discharged into a surge tank and thence exhausted to the trench. The entire system from

a potentiometer, and a needle rides directly over the adjustable band. This valve was used to pressurize the entire line to avoid vapor lock. Lubrication was obtained by feeding the vaporizing mask with clean [illegible].

CRITERION COOLING SYSTEM

Cooling was provided by circulating water through the water jacket in an externally driven circulating pump. The temperature was maintained somewhere at boiling point and evaporative cooling was promoted by cooling coils.

LUBRICATING SYSTEM

The oil pressure was obtained through an engine-driven pump. The oil circulating pump was externally driven. The oil temperature was [illegible] controlled by a heat exchanger using low pressure steam or water.

IGNITION SYSTEM

Ignition was provided by a [illegible] operating at approximately speed. The electrical circuit is shown in Fig. 5.

EXHAUST SYSTEM

The exhaust gases were discharged from the engine into a 1-1/4" iron pipe expanding into a 2 inch pipe in which the General Electric catalyst thermocouple was housed (Fig. 6). From here the gases were piped through a flexible steel coupling into a dust exchanger for cooling purposes. The gases from the dust exchanger discharged into a muffle tank and thence exhausted to the [illegible]. The entire system from

cylinder to discharge was lagged to reduce heat loss to a minimum. A detailed sketch of the exhaust system is shown in Fig. 3.

MEASURING INSTRUMENTS

A large electric cradle-dynamometer was used to measure torque (Fig. 7). Speed was determined by a calibrated mechanical tachometer. Air flow was measured with a standard sharp-edged orifice constructed in accordance with the ASME Fluid Meter Committee specifications. A differential water manometer was used to measure the pressure difference across the orifice. The orifice diameter was 0.515 inches with standard flange pressure taps. The calibration and corrections for temperature and pressure for this set-up are shown in Fig. 8. Fuel flow was measured with a Fischer and Porter Stabl-Vis Rotometer; calibration curves for this rotometer are shown in Fig. 9.

Atmospheric pressure was measured by a mercury barometer; inlet pressure was measured in the vaporizing tank by a water manometer, as was the exhaust pressure in the exhaust surge tank. Spark advance was measured by a neon light (excited by the spark discharge) which revolved with the crankshaft.

Cylinder inlet temperature was measured by a mercury bulb thermometer in the inlet pipe. Circulating water temperatures through the exhaust gas heat exchanger were measured by mercury bulb thermometers at intake and outlet. Exhaust gas temperatures were measured in four places; one, shortly after the exhaust port of the engine by the General Electric

cylinder to discharge was judged to produce heat loss to a minimum. A detailed sketch of the exhaust system is shown in Fig. 3.

EXPANSION INSTRUMENTS

A large electric overlie-dynamometer was used to measure torque (Fig. 7). Speed was determined by a calibrated mechanical tachometer. Air flow was measured with a standard sharp-edged orifice constructed in accordance with the ASME Fluid Meter Committee specifications. A differential water manometer was used to measure the pressure difference across the orifice. The orifice diameter was 0.515 inches with standard flange pressure taps. The calibration and correlations for temperatures and pressures for this setup are shown in Fig. 8. Fuel flow was measured with a Fleisher-Hoc Burke Stabil-Vis Rotameter; calibration curves for this rotameter are shown in Fig. 9.

Atmospheric pressure was measured by a mercury barometer; inlet pressure was measured in the receiving tank by a water manometer, as was the exhaust pressure in the exhaust surge tank. Spark advance was measured by a stone light (excited by the main distributor) which revolved with the crankshaft.

Cylinder inlet temperature was measured by a mercury bulb thermometer in the inlet pipe. Thermocouple water thermometers through the cylinder jackets were employed by mercury bulb thermometers at inlet and outlet. Exhaust gas temperatures were measured in four pieces one, shortly after the exhaust port of the engine by the special Chromel

shielded thermocouple, and at three places in the exhaust surge tank by iron-constantan thermocouples. Oil temperature was measured by an iron-constantan thermocouple in the crankcase. Fuel temperature was measured directly after the rotometer by a mercury bulb thermometer. The M.I.T. High Speed Indicator was used to obtain records of cylinder pressure versus crank angles.(Fig. 10). The M.I.T. transfer table was used to obtain pressure-volume diagrams therefrom.

solidified thermocouple, and at three places in the chamber surface back by iron-constantan thermocouples. Oil temperature was measured by an iron-constantan thermocouple in the crankcase. Final temperature was measured directly after the rotameter by a mercury in glass thermometer. The S.I.E. High Speed Indicator was used to obtain records of cylinder pressure versus crank angle.(Fig. 30). The S.I.E. trigger unit was used to obtain pressure-volume diagrams therefrom.

PROCEDURE

The primary consideration of this investigation was to correlate the results obtained from the three methods used to measure exhaust gas temperatures. This was done by taking simultaneous readings of all the data necessary, as shown in Table I. The runs made were divided into groups of three different compression ratios with five different fuel-air ratios for each compression ratio. Compression ratios used were 7, 8, and 9, and fuel-air ratios used were 80%, 90%, 100%, 110%, and 120% of chemically correct fuel-air ratios.

The following conditions were held at constant values as indicated:

1. Oil pressure 40 ± 2 psia.
2. Jacket water temperature 212°F
3. Inlet valve clearance .012"
4. Exhaust valve clearance .014"
5. Engine speed 1600 ± 10 rpm
6. Spark advance 20°

The following factors were varied as necessary for test:

1. Fuel-air ratio
2. Compression ratio

Many familiarization runs were made to determine the best operating conditions. During these familiarization runs many difficulties were discovered which had to be overcome in order to proceed. First, the heat exchanger, used to cool the exhaust gases, was much more efficient than originally

The primary consideration of this investigation was to correlate the results obtained from the three methods used to measure exhaust gas temperatures. This was done by tabulating simultaneous readings of all the data necessary, as shown in Table I. The runs made were divided into groups of three different compression ratios with five different fuel-air ratios for each compression ratio. Compression ratios used were 5, 6, and 7, and fuel-air ratios used were 80%, 90%, 100%, 110%, and 120% of chemically correct fuel-air ratios.

The following conditions were held at constant values as indicated:

1. Oil pressure	40 $\frac{1}{2}$ psia.
2. Jacket water temperature	212°F
3. Inlet valve clearance	.012"
4. Exhaust valve clearance	.018"
5. Engine speed	1800 $\frac{1}{2}$ rpm
6. Spark advance	30°

The following factors were varied as necessary for data:

1. Fuel-air ratio
2. Compression ratio

Many modifications were made in obtaining the best operating conditions. During these modifications many difficulties were discovered which had to be overcome in order to proceed. First, the best carburetor, used to meet the exhaust gases, was much more efficient than originally

designed (according to Ref. 4), and the cooler had to be decreased in size from 12' to 8', and from counter flow to parallel flow, to maintain a temperature well above 212°F in the exhaust surge tank. Also, due to excessive heat caused by lagging the exhaust pipes, a sodium-filled exhaust valve was used to maintain cooler valve temperature to lessen the probability of backfire.

desired (according to Ref. 9), and the cooler hot be by de-
creased in mine from 12° to 6°, and from counter flow to
parallel flow, to maintain a temperature well above 32°F
in the ethanol surge tank. Also, due to excessive heat
caused by lighting the exhaust lines, a bodine-fillet exhaust
valve was used to maintain cooler valve temperatures to insure
the probability of operation.

DISCUSSION

In this thesis three methods of exhaust gas temperature measurements were investigated. Two methods, namely, determination of temperature from the indicator diagram and from the heat exchanger required computation and use of the Hottel Burned Mixture Charts for (CH_2) (Fig. 11). The third method was the direct temperature reading of the General Electric Thermocouple. A description of the use of the Hottel charts and the high speed indicator diagram for the solution of this problem is given below.

To compute the exhaust gas temperature from a high speed indicator card the First Law of Thermodynamics was used. From a point "X" on the expansion stroke line immediately before the exhaust valve opens an equivalent cycle was used through the blowdown and exhaust processes.

It was necessary to find a relation between the cylinder volume and the Hottel Chart volume, and to determine the percentage of residual gas "f" in the actual cycle. These two must be done together by trial and error.

Relation between V_{chart} and $V_{cylinder}$.

In actual cycle:

Fresh air/cycle = B pounds

Air in residual gas/cycle = Z pounds

Total air/cycle = (B + Z) pounds

Total fuel/cycle = (B + Z)F pounds, where F = fuel-air ratio

Total charge/cycle = (B + Z)(1 + F) pounds

Total residual gas/cycle = Z(1 + F) pounds

Discussion

In this Sheila Otto method of exhaust gas temperature measurement, data investigated. The method, namely, determination of temperature from the indicator diagram and temperature exhaust manifold Sheila Otto (Obj) (Fig. 11). The third method was the direct temperature reading of the thermal electric thermocouple. A description of the use of the Bethel Otto and the high speed indicator diagram for the solution of this problem is given below.

To compute the exhaust gas temperature from a high speed indicator card the first law of thermodynamics was used. From a point "2" on the expansion stroke line immediately before the exhaust valve opens an adiabatic cycle was used through the blowdown and exhaust processes.

It was necessary to find a relation between the cylinder volume and the Bethel Otto volume, and to determine the percentage of residual gas "r" in the actual cycle. These two must be done together by trial and error.

Relation between Otto and Cylinder:

In actual cycle:

Press air/cycle = 3 pounds

Air in residual gas/cycle = r pounds

Total air/cycle = (3 + r) pounds

Total fuel/cycle = (3 + r)$\frac{1}{x}$ pounds, where $\frac{1}{x}$ = fuel-air ratio

Total charge/cycle = (3 + r)(1 + $\frac{1}{x}$) pounds

Total residual gas/cycle = r(1 + $\frac{1}{x}$) pounds

But $f = \frac{Z(1 + F)}{(B + Z)(1 + F)} = \frac{Z}{B + Z}$

So that $Z = \frac{Bf}{1-f}$

Total charge/cycle $= (B + Z)(1 + F) = (B + \frac{Bf}{1-f})(1+F) = \frac{B}{1-f}(1 + F)$

Chart mass $= (1 + F)$ pounds

Therefore $\frac{\text{chart mass}}{\text{cycle mass}} = \frac{1-f}{B}(\frac{1+F}{1+F}) = \frac{1-f}{B}$

And $\frac{V_{chart}}{V_{cylinder}} = \frac{1-f}{B}$

Thus a value for f was assumed and the $V_{cylinder}$ for point "X" converted to V_{chart}. This value and the pressure at point "X" were then used to locate the corresponding point on the Hottel Chart for the same fuel-air ratio.

Then expand adiabatically to the exhaust pressure to get the volume of the burned charge at exhaust pressure.

Now $f = \frac{V_{clearance}}{V_{exhaust}}$

This value of f was then compared with the assumed value, and the method of determining f repeated until the two values were equal.

Now the point "X" could be accurately located on the Hottel chart. By expanding to total cylinder volume converted to chart volume, and to exhaust pressure, two points "4" and "5" respectively, could be located on the Hottel chart.

Assuming that the chemical energy of the exhaust gas did not change during the exhaust process, the sensible enthalpy of the exhaust gas may be calculated as follows:

$$\text{But } f = \frac{(3+\gamma)(1+k)}{3(1+k)} = \frac{3+\gamma}{3}$$

$$\text{so that } k = \frac{f-1}{\frac{2\gamma}{3}}$$

$$\text{Total charge/cycle} = (3+\gamma)(1+k) = (3+\frac{2\gamma}{3}(1+k) = \frac{2-\gamma}{3}(1+k)$$

$$\text{Short mode} = (1+k) \text{ cycles}$$

$$\text{Therefore } \frac{\text{cycle time}}{\text{short mode}} = \frac{3}{1-f(\frac{1+k}{3})} = \frac{3}{1-f}$$

$$\text{and } \frac{\lambda_{\text{elliptic}}}{\lambda_{\text{short}}} = \frac{3}{1-\gamma}$$

Thus a value for γ only assumed and the $\lambda_{\text{elliptic}}$ for point "X" converted to λ_{short}. This value and the pressure at point "X" were given would be inside the corresponding point on the Mollier Chart for the same fuel-air ratio.

Then expand volumetrically to the values pressure in the volume of the burned charge at ambient pressure.

$$\text{Now } r = \frac{\lambda_{\text{polytropic}}}{\lambda_{\text{ambient}}}$$

This value of r was then compared with the "assumed value" and the method of determining γ repeated until the two values were equal.

See the point "X" could be accurately located on the Mollier chart. By expanding to total cylinder volume compared to clear volume, and to ambient pressure, two points "P" and "S" respectively, could be located on the Mollier chart.

Assuming that the constant energy of the exhaust gas did not change during the exhaust process, the reversible adiabatic of the exhaust gas may be calculated as follows:

$$(1-f)H_{se} = E_{s4} - fE_{s5} + \frac{P_e(V_1-V_2)}{J}$$

and the temperature of the exhaust gas determined from the Hottel chart.

Determination of Exhaust Gas Temperature by the Heat Extraction Method

The heat extracted from exhaust gases (Btu/sec) by the water was found from the rate of water flow (lbs/sec) and its rise in temperature. Since the basis of the Hottel charts is one pound of air plus the necessary fuel to give the proper fuel air ratio, the Btu's/sec found from the water flow was converted to Btu's/lb air by dividing this value by the mass rate flow of air in lbs/sec.

Using the mean temperature of the gases in the surge tank, the enthalpy of these gases was determined from the Hottel chart at an entropy of 0.30. This entropy was used because the actual point was off the chart. This introduced no error because in this region enthalpy is a function of temperature only. By adding the heat extracted (Btu/lb air) the enthalpy of the gases at exhaust was found. From this value the temperature was read directly.

Design Considerations

The design of the heat exchanger for cooling was critical. The area had to be such that a reasonable rise in water temperature was produced (to minimize errors in reading); that the rate of water flow was sufficient to prevent

$$(T_2)^{1.5} = \frac{T_1^{1.5} - kC_p^{0.5} + \frac{kC_p^{0.5}(\Delta T_1 - \Delta T_2)}{2}}{}$$

and the temperature of the exhaust gas determined from the Mollier chart.

Determination of Exhaust Gas Temperature by the Heat Calorimeter Method

The heat extracted from exhaust gases (BTU/sec) by the water was found from the rate of water flow (lbm/sec) and its rise in temperature. Since the basis of the Mollier chart is one pound of air plus the necessary fuel to give the proper fuel air ratio, the gas's mass flow was converted to gas/lb air by dividing this value by the mass ratio flow of air to lbm/sec.

Using the mean temperature of the gases in the engine jacket, the enthalpy of these gases was determined from the Mollier chart at an entropy of 0.30. This entropy was used because the actual point was off the chart. This interpolation be error because in this region enthalpy is a function of temperature only. By reading the heat extracted (BTU/lb air) the relative to the gases at exhaust was found. From this value the temperature was read directly.

Design Considerations

The design of the heat exchanger for cooling the exhaust; that the heat of water flow was sufficient to prevent water temperatures was produced (to minimize errors in readings) the gas to be cooled must a reasonable rate to

steam pockets; and that the temperature of the exhaust gases in the surge tank was sufficiently high to avoid water condensation.

Three ninety degree turns were made in the exhaust system before the heat exchanger. This was deemed necessary in order to prevent the shielded thermocouple from "seeing" the hot exhaust port or the entrance to the cooler. Also, these turns induced turbulence and helped to prevent stratification of the exhaust gases. To minimize the possibility of a large temperature gradient across the thermocouple, a third turn was introduced immediately before the cooler. This turn directed the exhaust gases vertically upward into the cooler which allowed easy removal of air from the cooling water system.

The heat exchanger consisted of a two-inch pipe concentrically placed in a three-inch pipe. One bend was made in the cooler making the discharge horizontal for convenience of installation and instrumentation. This was done by spider spacing around the inner pipe, tightly packing sand between the pipes, and cold bending the combination. This method proved very satisfactory and is recommended for any similar construction.

The surge tank was used to bring the exhaust gases to a sufficiently stagnant condition to allow accurate pressure and temperature measurements.

The whole system was lagged with high temperature insulation and it very effectively minimized heat losses to the

lation and it was originally structured just above the
water condensation.

The whole system was lighted with high temperature lamps
and incandescent instruments.

The large tank was used to store the mineral fluids to
a sufficiently elevated temperature to allow convective circulation
and circulate constantly.

This method proved very satisfactory but it consumed far
too much power. The lights, and only heating the mechanisms.
Some of mineral content crossed into the water tube,
clogging the system, and only heating the mechanisms. This was
done by mixing cooling crossed into the water tube, slightly decreasing
the temperature of insulation and insulation. This was
made in the cooler making the discharge horizontal for
convenience of installation and insulation. This was
successfully placed in a three-tank system. The heat was
mainly removed at the top from the cooling water system.

The heat conductor consisted of a low-loss tube
surrounded by the condenser. Each tube discharged the con-
densed fluids continually cooled into the cooler again allowed
more removal at the top from the cooling water system.

The heat conductor consisted of a low-loss tube
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the heat conductor consisted of a low-loss tube
surrounded by the condenser. Each tube discharged the con-
densed fluids continually cooled into the cooler again allowed
more removal at the top from the cooling water system.

atmosphere. Due to this lagging, it is recommended that the exhaust pipes and flexible coupling prior to the heat exchanger be made of high temperature steel, if the use of a measuring thermocouple is anticipated. The use of a flexible coupling between the exhaust pipe and heat exchanger was considered advisable in this investigation to prevent damage to the engine cylinder head due to vibration between the engine mount and the heavy cooling system, and to allow for changing of the compression ratio without moving the heat exchanger.

allowance. Due to this finding, it is recommended that the minimum piston ring flexible coupling prior to the bent passenger be made of high temperature steel, if the use of a measuring thermocouple is anticipated. The use of a flexible coupling between the exhaust pipe and heat exchanger was considered critical in this investigation to prevent damage to the engine cylinder head due to vibration between the engine mount and the heavy cooling system, and to allow the shooting at the observation ports without moving the heat exchanger.

RESULTS

The temperatures were computed according to the methods described in the Discussion and were plotted in Figs. 12-17. As can be seen from the curves, the trend of temperature change by each method agreed exceedingly well. Since a spark advance of 20 degrees was used in all runs, the shape of the curves for each set of compression ratios differed from one another. This was apparently due to the fact that best power spark advance was not used for each compression ratio and fuel air ratio.

The results from the heat exchanger method used in this investigation were not quite in agreement with the direct reading of the G.E. thermocouple and were well out of agreement with the indicator method. This was expected due to the simplifying assumptions made, namely, that the exhaust system process was adiabatic and the "blow-down" was isentropic.

The temperatures computed by the indicator method were fundamentally theoretical and are not to be considered the actual temperatures existing in the exhaust pipe. However, the nearer to this temperature that any method can come, the better the method. From the curves, the temperature difference between the theoretical and the computed temperature is on the order of 400°F. This is not considered to be close enough to give an accurate heat balance of the engine, nor to predict accurately the

RESULTS

The temperatures were evaluated according to the methods described in the Discussion and were plotted in Figs. 12-17. It can be seen from the curves, the trend of temperature change is very nearly exponentially well true, the choice of the curves for each set of combustion ratios differed from one another. This was primarily due to the fact that heat power eight advance was not used for rough compression tests and two air ratio.

The results from the heat evaporator method in this investigation were not quite in agreement with the direct reading of the O.S. thermocouple and were only one of agreement with the indicator method. This was supposed due to the simplifying assumptions made, namely, that the exhaust system process was adiabatic and the "plug-flow" was investigate.

The temperatures computed by the indicator method were fundamentally theoretical and are not to be considered the actual temperatures existing in the exhaust pipe. However, the nearer to this temperature that any method can come, the better the method. From the nature, the temperature differences between the theoretical and the computed temperature is on the order of 100°. This is not considered to be close enough to give an accurate

energy remaining in the exhaust gases. No accurate heat balance has ever been performed on an internal combustion engine, but according to theoretical considerations, approximately 15% to 20% of the heating value of the fuel is dissipated in the cylinder and jacket. Because the exhaust pipe and port in this investigation were at such a high temperature, due to the lagging, it was expected that the heat transfer back to the cylinder from the exhaust pipe would appreciably increase the amount indicated above. Therefore, by taking an estimated 8% to 11% of the heating value of the fuel, additionally dissipated to the cylinder and jacket, and applying this correction to the curves already obtained by the cooler method, they can be made to fall directly on the indicator curves.

The G.E. thermocouple gave results that were closer to the indicator temperatures, the average error from this temperature being about 6% to 7% of the heating value of the fuel. Because of its small size and simple installation, this thermocouple was considered both practical and accurate. Therefore, it is concluded that this method is, to date, the most satisfactory method of measuring high temperatures in a hot moving stream within the limits of the instrument, that is, up to 1800 degrees Fahrenheit.

Since the error between the G.E. thermocouple reading and the heat exchanger method is only about three percent of the heating value of the fuel, and since this error can

be further reduced or eliminated by connecting the heat exchanger, with an insulating coupling, directly to the exhaust port of the engine, the method developed in this thesis is recommended for use with engines whose exhaust temperatures are both in and above the temperature range of the G.E. thermocouple. This is based on the assumption that the G.E. thermocouple gave results closest to the actual conditions.

be further reduced or eliminated by connecting the heat exchanger, with an insulating coupling, directly to the exhaust port of the engine, the vented developer in this device is recommended for use with engines whose exhaust temperatures are both in and above the temperature range of the S.E. thermocouple. This is based on the assumption that the S.E. thermocouple gave results closest to the actual conditions.

CONCLUSIONS

1. Satisfactory correlation was obtained among the three methods investigated.

2. The General Electric shielded thermocouple gave values closest to the expected results and is the most practical reliable method.

3. The heat exchanger method is sufficiently accurate for the measurement of temperatures above the range of the G.E. thermocouple.

4. The indicator method can be used for rough results by subtracting approximately 6% of the heating value of the fuel from the energy of the exhaust gases determined by this method.

CONCLUSIONS

1. Satisfactory correlation was obtained among the three methods investigated.

2. The Dewrad Electric calorimeter gave values closest to the accepted results and is the most practical reliable method.

3. The best economical method is multiplexing reactors for measurement of temperatures above the range of the D.E. thermocouple.

4. The indicator method can be used for rough results by multiplexing approximately 6% of the heating value of the fuel from the energy of the exhaust given thereby by this method.

RECOMMENDATIONS

1. This investigation should be further carried on, connecting the water cooler (by insulated coupling), directly to the exhaust port to further reduce heat losses.
2. To establish a definite trend of the temperature curves, best power spark advance should be used for all runs.
3. If a similar set-up is used, high temperature metals should be used for the portion of the exhaust pipe between engine and cooler.

CONCLUSIONS

1. The investigation should be further carried on, modelling the water cooler (by insulated cooling), directly to the cathode port to further reduce back losses.

2. To establish a definite trend of the temperature curve, pure power ejects stages should be used for all runs.

3. If a similar set-up is used, high temperature metals should be used for the portion of the element plus between cooling and cooler.

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2. "Fluid Meters, Their Theory and Application", Part I, ASME Research Publication, 4th Edition, 1937.

3. "ASME Power Test Codes of 1940", Part 5, Chapter 4, Flow Measurement, July 1940.

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2. "Fluid Meters, Their Theory and Application", Part I, ASME Research Publications, 4th Edition, 1937.

3. "ASME Power Test Codes of 1940", Part 5, Chapter 4, Flow Measurements, ASME 1940.

4. Wickham, W. E.: "Heat Transmission", 2nd Edition, Hodges, Hill Book Company, 1942.

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6. "Temperatures - Its Measurement and Control in Science and Industry", National Publishing Corp., 1941.

SAMPLE CALCULATIONS

Heat Exchanger Method

Conditions: Compression ratio 8
Fuel air ratio 120% of chemically correct

Mean temperature in exhaust surge tank (°F)	303
Mass rate of water flow (lbs/min)	3.43
" " " air " (lbs/sec)	0.0149
Rise in cooling water temperature (°F)	104

Heat extracted by cooling water = $\frac{3.43 \times 104}{60}$ = 5.95 Btu/sec

= $\frac{5.95}{.0149}$ = 399 Btu/lb air

Sensible enthalpy of gases @ 303°F (Btu/lb air)	102
Rise in sensible enthalpy of gases (Btu/lb air)	399
Final sensible enthalpy of gases (Btu/lb air)	501
Temperature of exhaust gases at exhaust (°F)	1555

Indicator Card Method

Conditions: Compression ratio 8
Fuel air ratio 120% of chemically correct

Symbols:
- M_a - mass of air (lbs/cycle)
- V_{cyl} - actual cylinder volume (cubic inches)
- f - mass fraction of residual gases
- P_x - pressure at point "x" on indicator card
- V_x - cylinder volume at point "x" on indicator card.
- E_s - sensible internal energy (Btu/lb products)
- H_s - sensible enthalpy (Btu/lb products)

$M_a = \frac{\text{lbs/sec}}{\text{rps}/2} = \frac{.0149}{1593} \times 120 = .001121$

Clearance volume = 5.333 cubic inches

Assume f = .044

Heat Exchanger Design

Conditions: Compression ratio 8
Total air ratio 130% of stoichiometric

Mean temperature in exhaust outlet bank (^{o}F)	393
Mean rate of water flow (lbm\min)	3.43
" " " air " (lbm\sec)	6.0149
Mean in cooling water temperature (^{o}F)	158

Heat extracted by cooling water = $\frac{3.43 \times 158}{60}$ = 3.93 Btu\sec

= $\frac{3.93}{.0149}$ = 399 Btu\lb air

Possible enthalpy of gases = 393^{o}F [Btu\lb air]	102
Mine in exhaust portion of gases [Btu\lb air]	399
Ideal maximum enthalpy of cases [Btu\lb air]	501
Temperature of exhaust gases at entrance (^{o}F)	1215

Jet/Water Cool Design

Conditions: Compression ratio 8
Total air ratio 130% of stoichiometric

Symbols: E_a - mass of air (lbm\stroke)

V_{cyl} - actual cylinder volume (metric tonnes)

γ - mass fraction of residual gases

p_x - pressure at point "x" on indicator card

V_x - cylinder volume at point "x" on indicator card.

E_e - available internal energy (Btu\lb products)

E_a - available enthalpy (Btu\lb products)

E_r = $\frac{lbm\sec}{lbm\sec}$ = $\frac{1393}{.9242}$ x 130 = .001321

Clearance volume = 3.333 cubic inches

Ratio $\frac{L}{r}$ = .044

From Indicator diagram

$$P_x = 65.95\ \#/in^2 \qquad V_x = 37.73\ in^3$$

$$V_{chart} = \frac{37.73}{1728} \times \frac{.956}{.001121} = 18.6\ ft^3$$

$$V_{clearance} = \frac{5.333}{1728} \times \frac{.956}{.001121} = 2.63\ ft^3$$

From Chart

$$V_5 = 60\ ft^3 \qquad \therefore f = \frac{V_{clearance}}{V_5} = \frac{2.63}{60} = .044$$

From Indicator diagram

$$V_4 = 42.664\ in^3$$

$$V_4 = \frac{42.664}{1728} \times \frac{.956}{.001121} = 21.05$$

From Chart

$$E_{s4} = 885 - 336 = 549$$

$$E_{s5} = 692 - 336 = 356$$

$$(1 - f)H_{se} = E_{s4} - fE_{s5} + \frac{P_e(V_1 - V_2)}{J}$$

$$= 549 - .044 \times 356 + \frac{14.7 \times 144 \times 18.32}{778}$$

$$= 583.6$$

$$H_{se} = 610\ \text{Btu/\# air + fuel}$$

$$T_e = 2345^\circ R = 1885^\circ F$$

From Indicator diagram

$$P_3 = 63.93 \text{ lb/in}^2 \qquad V_3 = 37.73 \text{ in}^3$$

$$V_{clearance} = \frac{37.73}{1728} \times \frac{.0921}{.0921} = 18.6 \text{ in}^3$$

$$V_{clearance} = \frac{3.139}{1728} \times \frac{.0921}{.0921} = 8.63 \text{ in}^3$$

From Chart

$$P_1 = 60 \text{ lb/in}^2 \qquad \therefore f = \frac{V_{clearance}}{V_1} = \frac{8.63}{18} = .044$$

From Indicator diagram

$$V_4 = 42.664 \text{ in}^3$$

$$V_4 = \frac{42.664}{1728} \times \frac{.0921}{.0921} = 21.05$$

From Chart

$$E_{34} = 895 - 336 = 349$$

$$E_{45} = 692 - 336 = 356$$

$$(1 - f)H_{ad} = E_{ad} - E_{45} + \frac{P_{ad}(V_3 - V_4)}{2}$$

$$= 349 - .044 \times 356 + \frac{36.9 \times 144}{778} = 16.32$$

$$= 983.6$$

$$E_{ad} = 970 \text{ BTU/lb} \text{ air} + P_{ad}$$

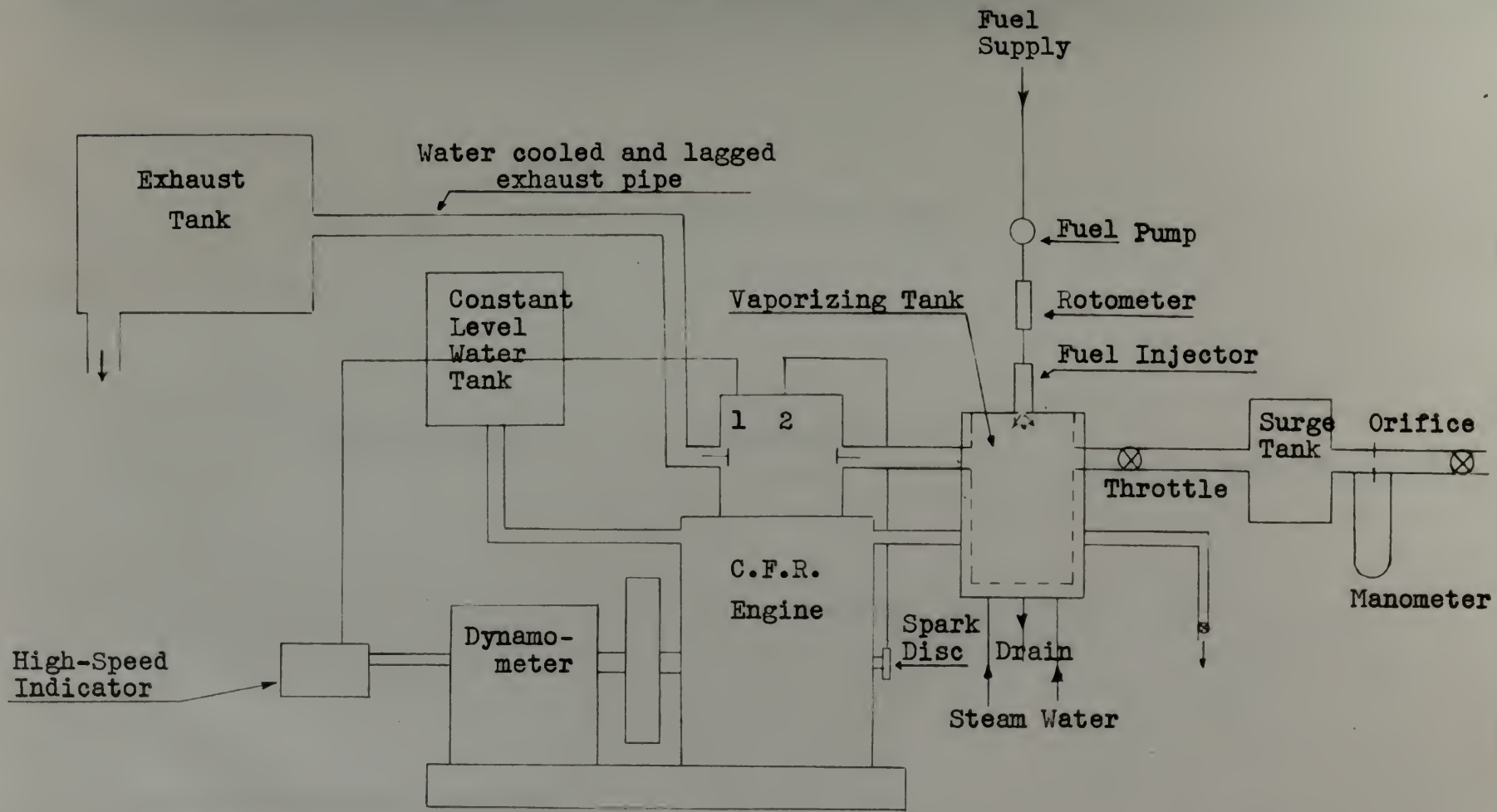
$$E_a = 1343 P_a + 1289 P_a$$

SLOAN LABORATORY

ENGINE CFR BORE 3.25" STROKE 4.5"

REMARKS	DATE	TIME	Run	RPM	B.L.	COMP RATIO	OIL PRES.	TEMP OIL	TEMP JACK.	P_i	P_E	T_i	AIR CONS.	FUEL CONS.	F/A	S.A.	AIR P_i	AIR ΔP	FUEL ROTO.	WATER T_i	WATER T_o	WATER W	THERMOCOUPLES 1	2	3	G.E.
					"Hg		PSI	°F	°F	"Hg	"Hg	°F	#/SEC.	#/SEC.		°	"Hg	"H_2O	85°F	°F	°F	#/MIN.	°F	°F	°F	°F
	4/12	1600	1	1610	7.25	7	40	160	212	28.30	29.92	141	.0150	.00120	.0798	20	29.92	12.30	9.90	54	170	2.95	330	325	337	1643
			2	1610	7.28	7	40	160	212	28.30	29.92	140	.0148	.00108	.0731	20	29.92	12.00	9.34	54	163	3.33	304	302	314	1744
			3	1610	7.10	7	40	160	212	28.30	29.92	155	.0148	.000993	.0665	20	29.92	12.00	8.85	56	174	3.38	328	323	338	1840
			4	1610	6.50	7	40	160	212	28.30	29.92	155	.0148	.000886	.0599	20	29.92	12.00	8.37	56	170	3.56	330	330	340	1820
			5	1610	5.60	7	40	160	212	28.30	29.92	155	.0148	.00079	.0532	20	29.92	12.00	7.85	57	173	3.51	338	332	348	1864
	4/24	1700	6	1593	6.95	8	38	136	212	28.11	29.73	150	.0149	.00119	.0798	20	29.73	12.14	9.85	56	160	3.43	300	300	310	1626
			7	1590	6.85	8	38	136	212	28.11	29.73	150	.0149	.00109	.0731	20	29.73	12.14	9.37	56	155	3.67	300	300	310	1705
			8	1587	6.60	8	38	134	212	28.11	29.73	150	.0149	.000991	.0665	20	29.73	12.14	8.90	56	157	3.71	301	301	311	1712
			9	1590	6.25	8	39	132	212	28.11	29.73	150	.0154	.000921	.0599	20	29.73	13.00	8.55	56	162	3.47	300	300	310	1706
			10	1590	5.40	8	39	129	212	28.11	29.73	150	.0156	.00083	.0532	20	29.73	13.30	8.07	56	175	3.09	303	302	313	1733
	4/24	2200	11	1595	8.60	9	41	126	212	28.11	29.73	134	.0156	.001249	.0798	20	29.73	13.30	10.08	55	162	2.90	279	278	288	1457
			12	1594	8.60	9	42	126	212	28.11	29.73	134	.0155	.001134	.0731	20	29.73	13.20	9.58	55	166	2.92	286	286	297	1546
			13	1594	8.50	9	42	125	212	28.11	29.73	129	.0156	.001039	.0665	20	29.73	13.30	9.14	55	176	2.80	300	299	310	1605
			14	1587	8.10	9	42	125	212	28.11	29.73	129	.0156	.000935	.0599	20	29.73	13.30	8.61	55	160	3.20	300	300	310	1565
			15	1577	8.10	9	42	125	212	28.11	29.73	129	.0156	.00083	.0532	20	29.73	13.30	8.06	54	158	3.25	299	299	309	1577

TABLE I



1. Indicator Cylinder Attachment
2. Spark Plug

SLOAN LABORATORY
EXPERIMENTAL C.F.R. ENGINE SETUP

Fig. 1

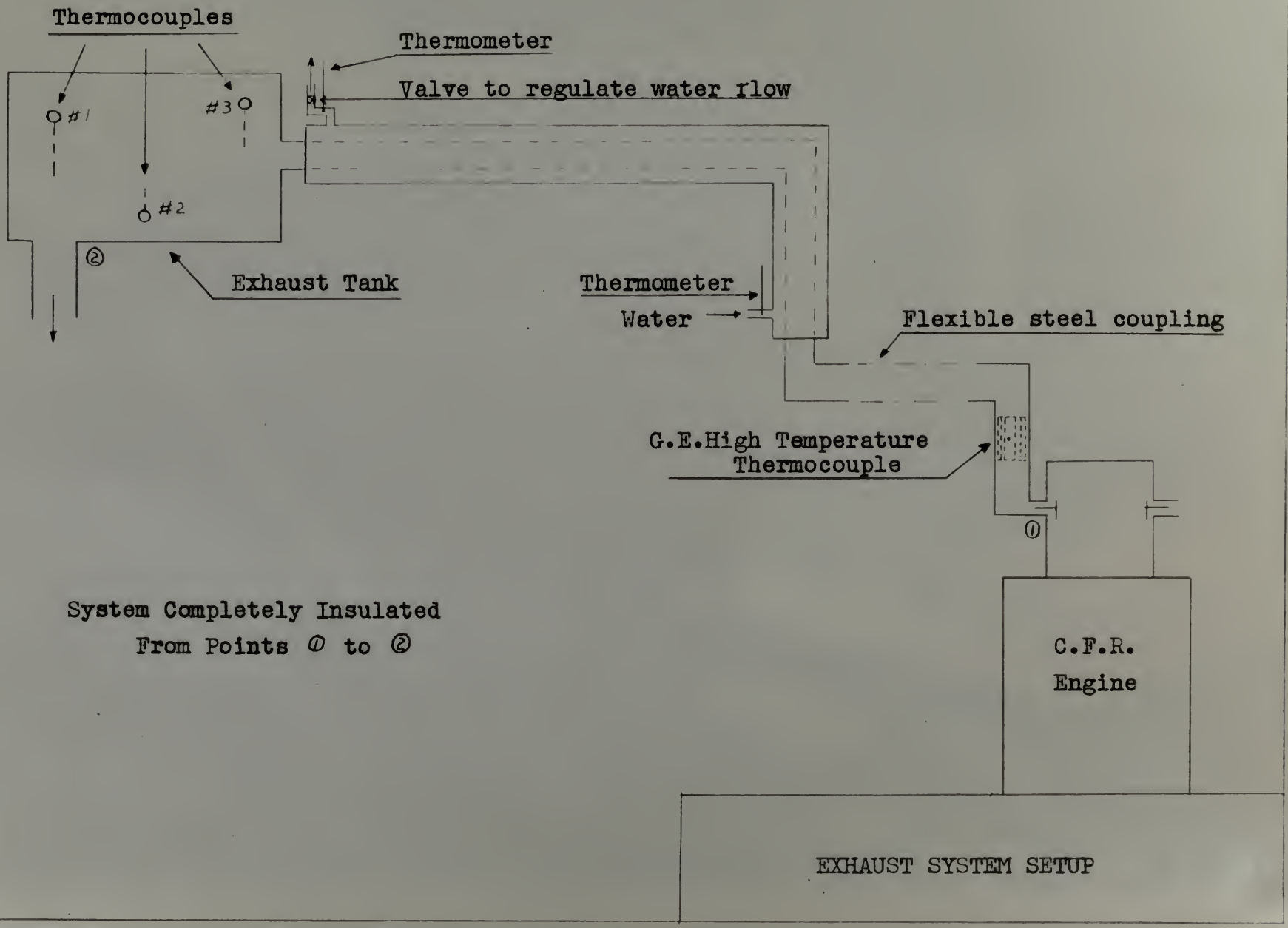
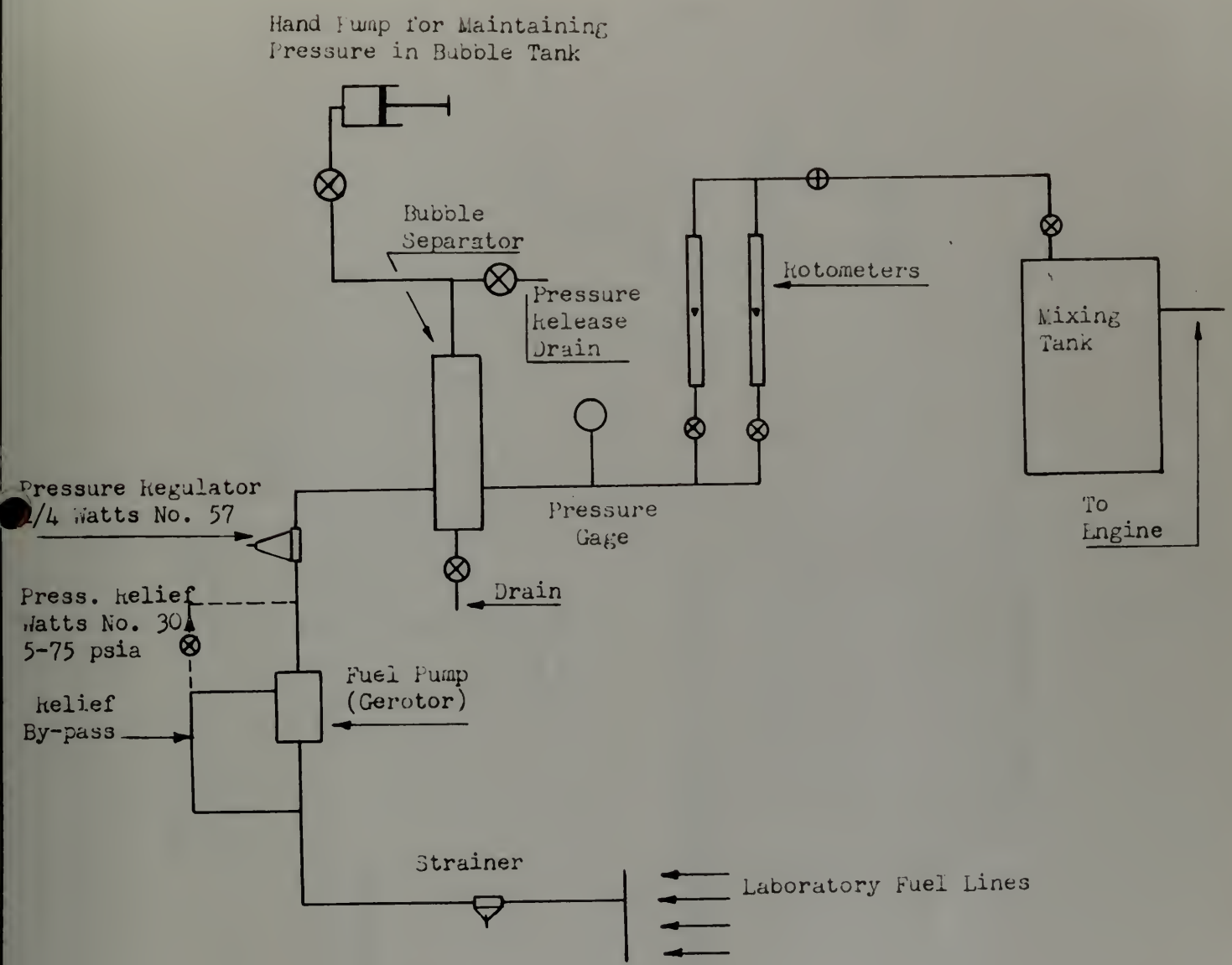

Thermocouples
Thermometer
Valve to regulate water flow
#1
#3
#2
②
Exhaust Tank
Thermometer
Water
Flexible steel coupling
G.E.High Temperature
Thermocouple
①
System Completely Insulated
From Points ① to ②
C.F.R.
Engine
EXHAUST SYSTEM SETUP

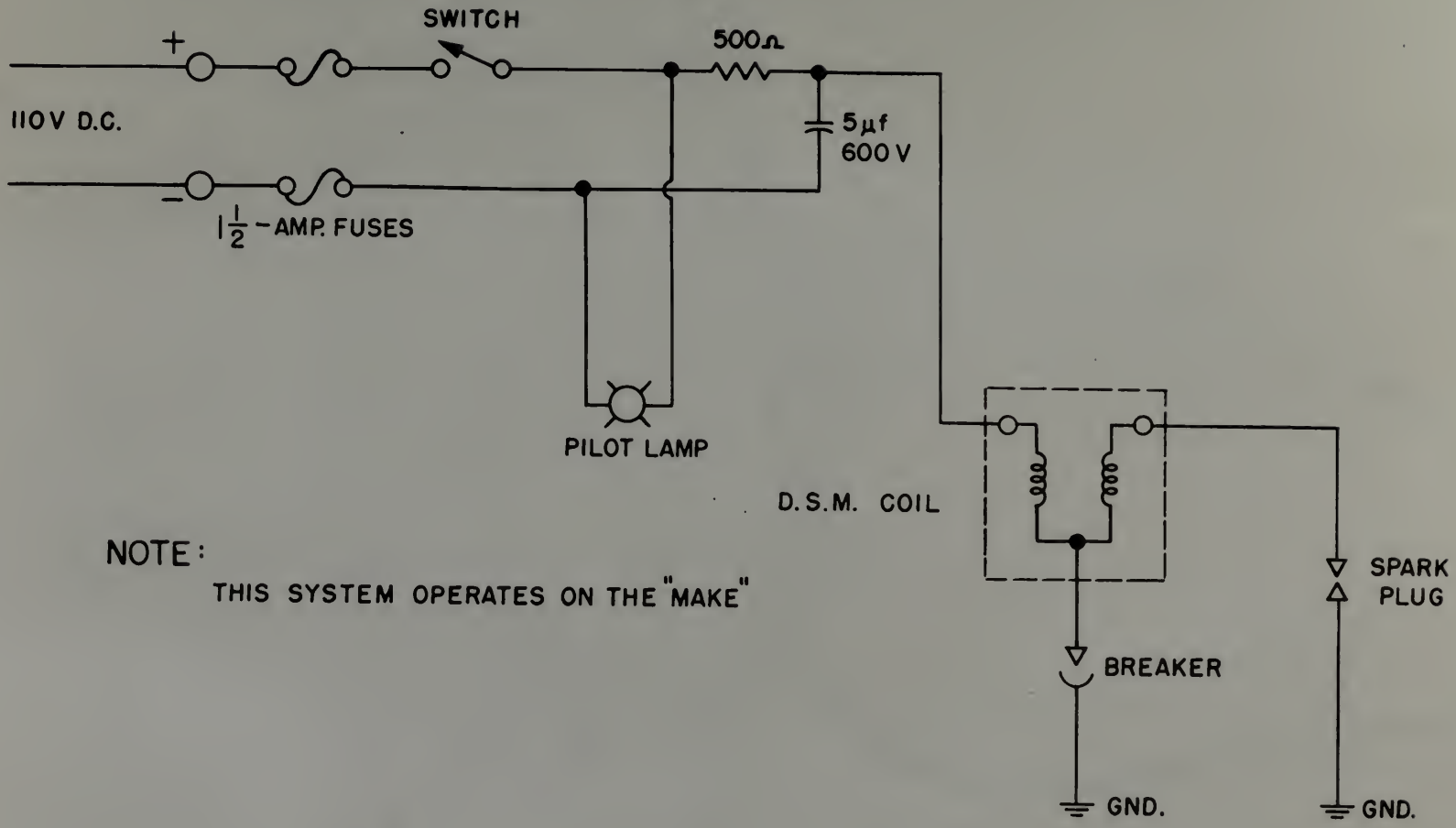
Fig. 2

Fig 3



SCHEMATIC DIAGRAM OF FUEL SYSTEM

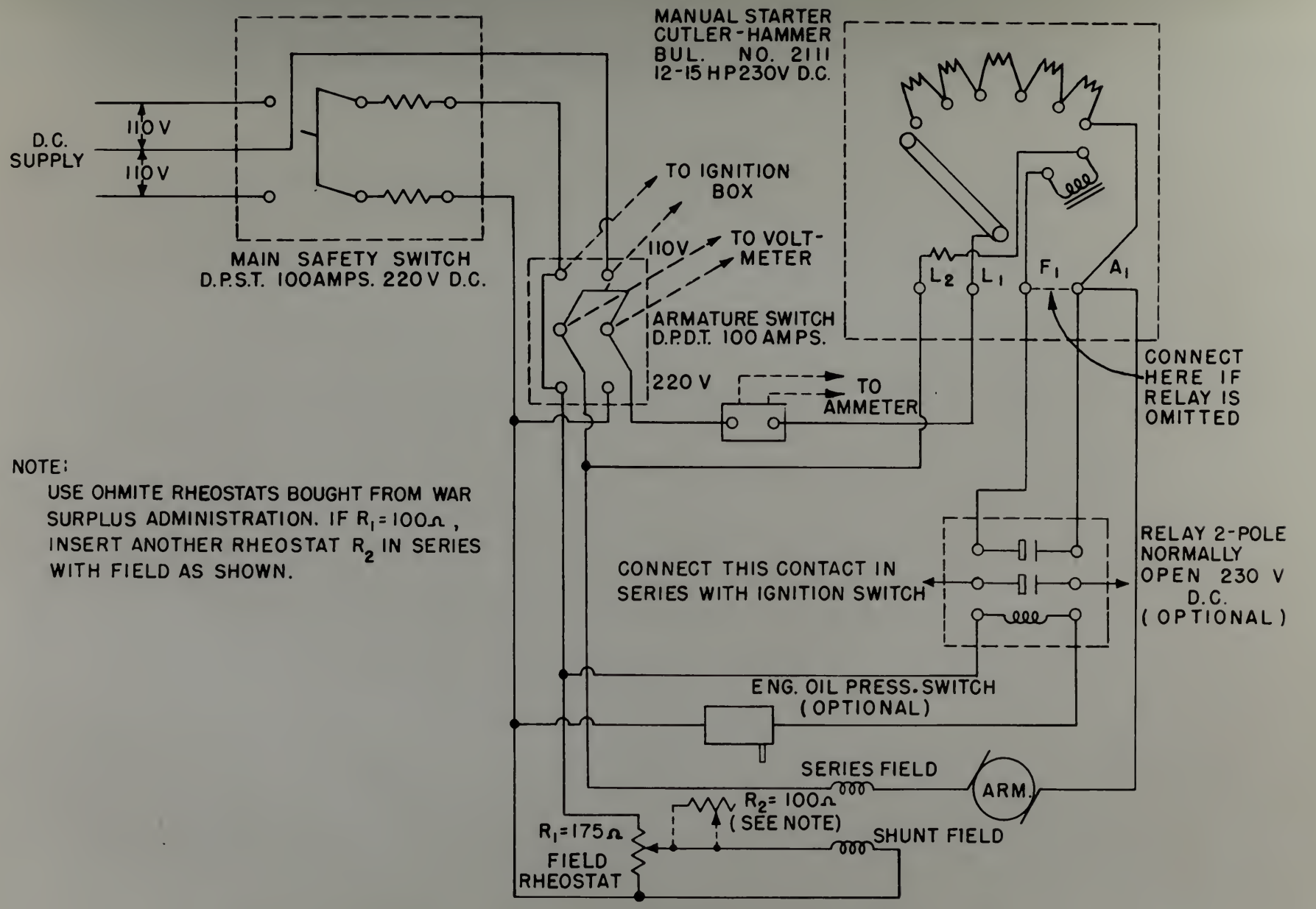
Fig.4



WIRING DIAGRAM OF IGNITION SYSTEM

FIG. 5

FIG. 6



WIRING DIAGRAM OF STAR DYNAMOMETER – CFR ENGINE SETUP

FIG. 7

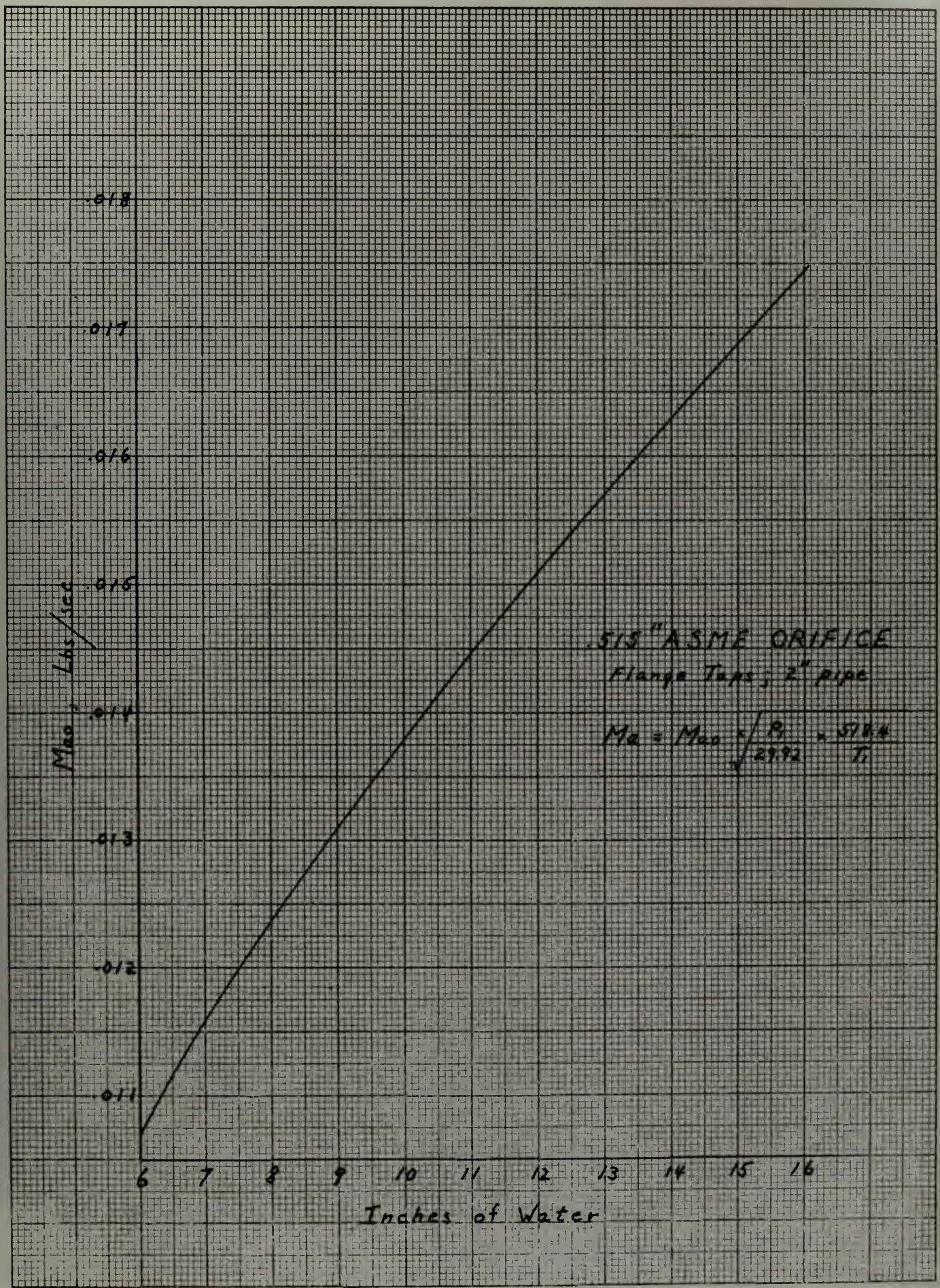

.018
.017
.016
.015
.014
.013
.012
.011
Mao - Lbs/sec
.515" ASME ORIFICE
Flange Taps ; 2" pipe
$Ma = Mao \times \sqrt{\frac{P_1}{29.92}} \times \frac{518.4}{T_1}$
6
7
8
9
10
11
12
13
14
15
16
Inches of Water

FIG. 8

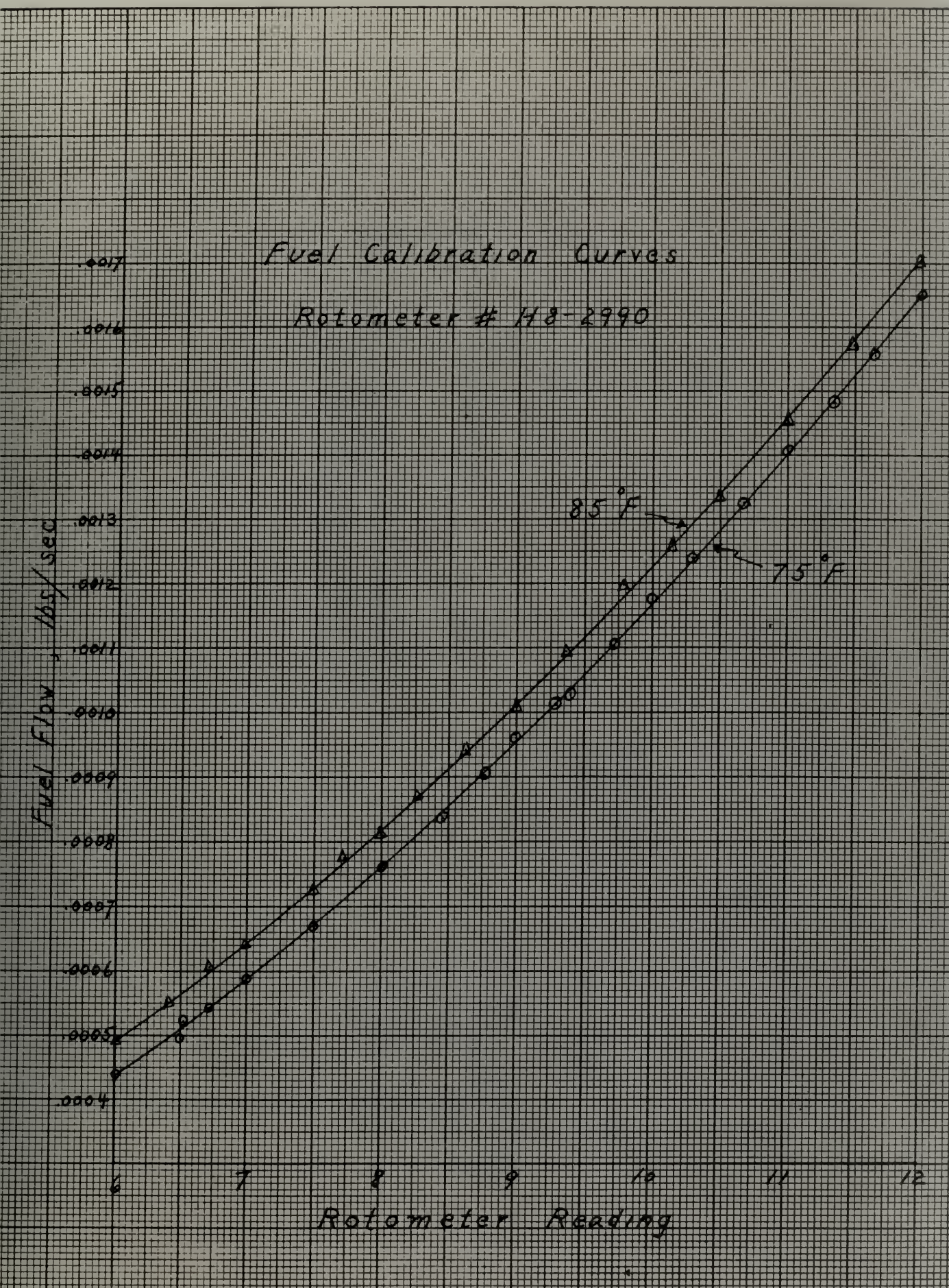
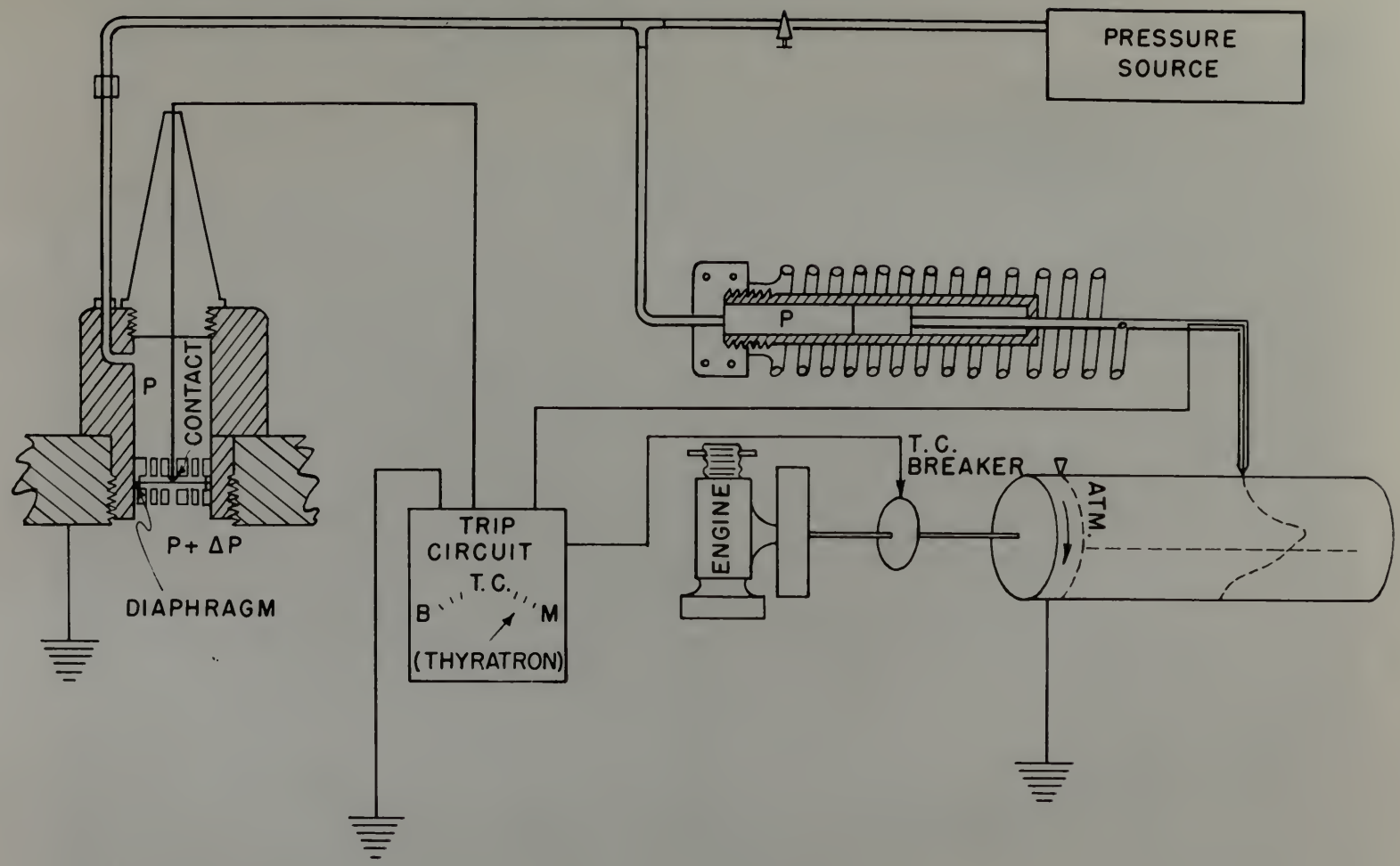

Fuel Calibration Curves
Rotometer # 118-2990
85°F
75°F
Fuel Flow, lbs/sec
.0017
.0016
.0015
.0014
.0013
.0012
.0011
.0010
.0009
.0008
.0007
.0006
.0005
.0004
6
7
8
9
10
11
12
Rotometer Reading

Fig. 9



HIGH SPEED INDICATOR

M. I. T.

Fig. 10

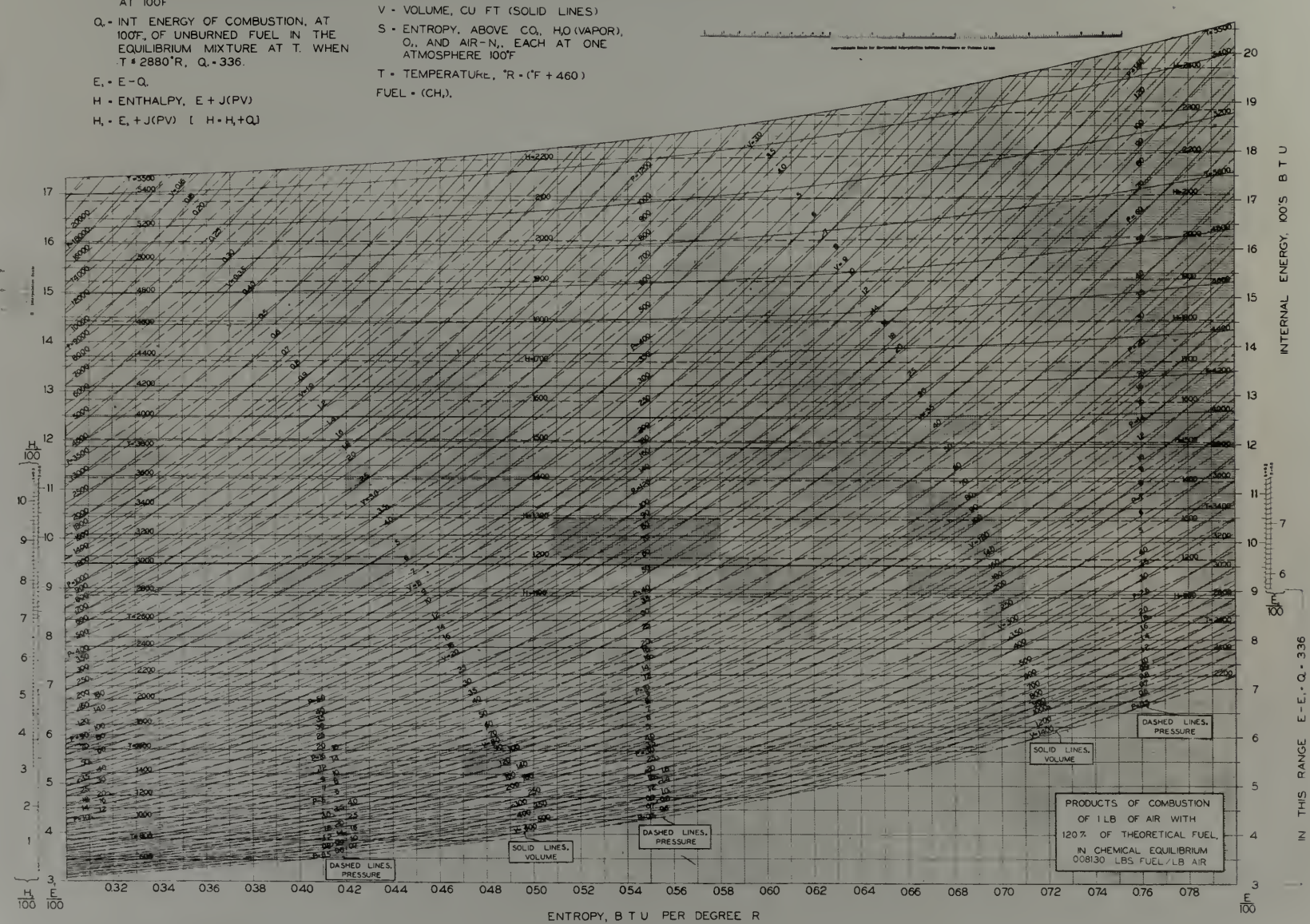

E = INTERNAL ENERGY, B T U, ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2 AT 100°F
Q = INT ENERGY OF COMBUSTION, AT 100°F, OF UNBURNED FUEL IN THE EQUILIBRIUM MIXTURE AT T, WHEN T = 2880°R, Q = 336.
$E_1 = E - Q$.
H = ENTHALPY, $E + J(PV)$
$H_1 = E_1 + J(PV)$ [$H = H_1 + Q$]
P = PRESSURE, LBS / SQ IN (DASHED LINES)
V = VOLUME, CU FT (SOLID LINES)
S = ENTROPY, ABOVE CO_2, H_2O (VAPOR), O_2, AND AIR – N_2, EACH AT ONE ATMOSPHERE 100°F
T = TEMPERATURE, °R = (°F + 460)
FUEL = $(CH_4)_x$
INTERNAL ENERGY, 100'S B T U
ENTROPY, B T U PER DEGREE R
IN THIS RANGE E – E_1 = Q = 336
DASHED LINES, PRESSURE
SOLID LINES, VOLUME
PRODUCTS OF COMBUSTION OF 1 LB OF AIR WITH 120% OF THEORETICAL FUEL, IN CHEMICAL EQUILIBRIUM
0.08130 LBS FUEL / LB AIR
M I T CAMBRIDGE, MASS MAY 1943

FIG. 11

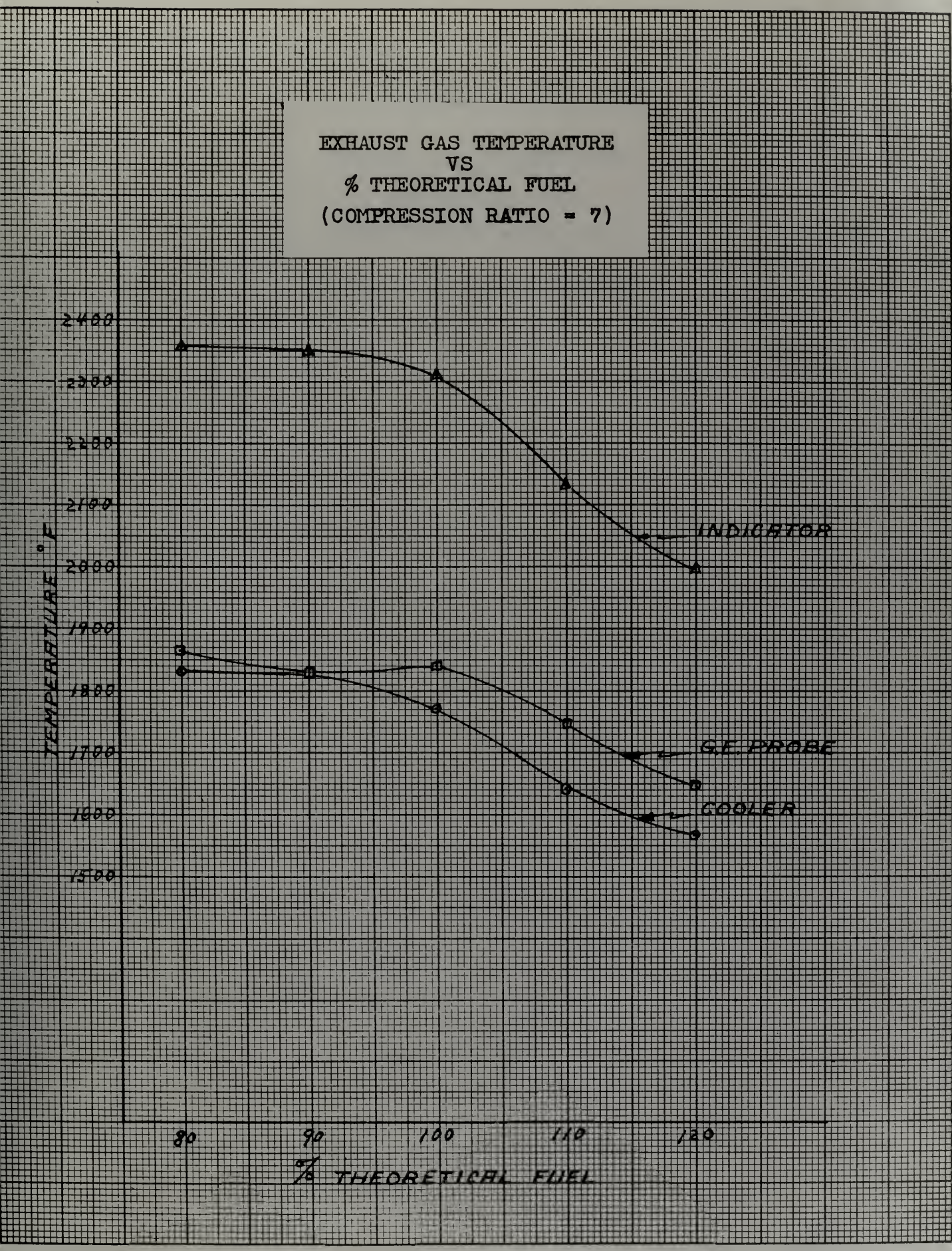

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COMPRESSION RATIO = 7)
TEMPERATURE °F
2400
2300
2200
2100
2000
1900
1800
1700
1600
1500
INDICATOR
G.E. PROBE
COOLER
80
90
100
110
120
% THEORETICAL FUEL

FIG. 12

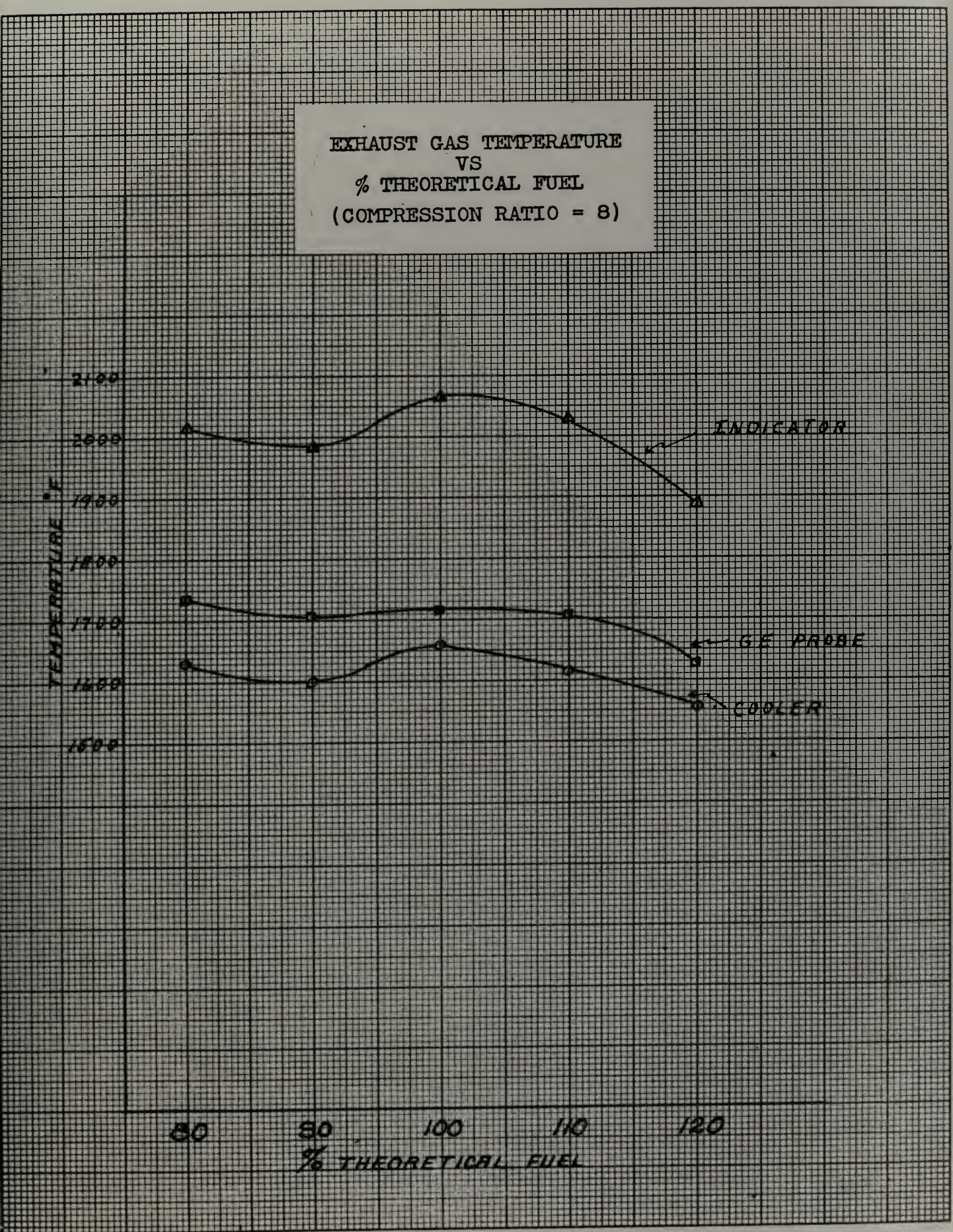

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COMPRESSION RATIO = 8)
TEMPERATURE °F
2100
2000
1900
1800
1700
1600
1500
INDICATOR
GE PROBE
COOLER
80
90
100
110
120
% THEORETICAL FUEL

FIG. 13

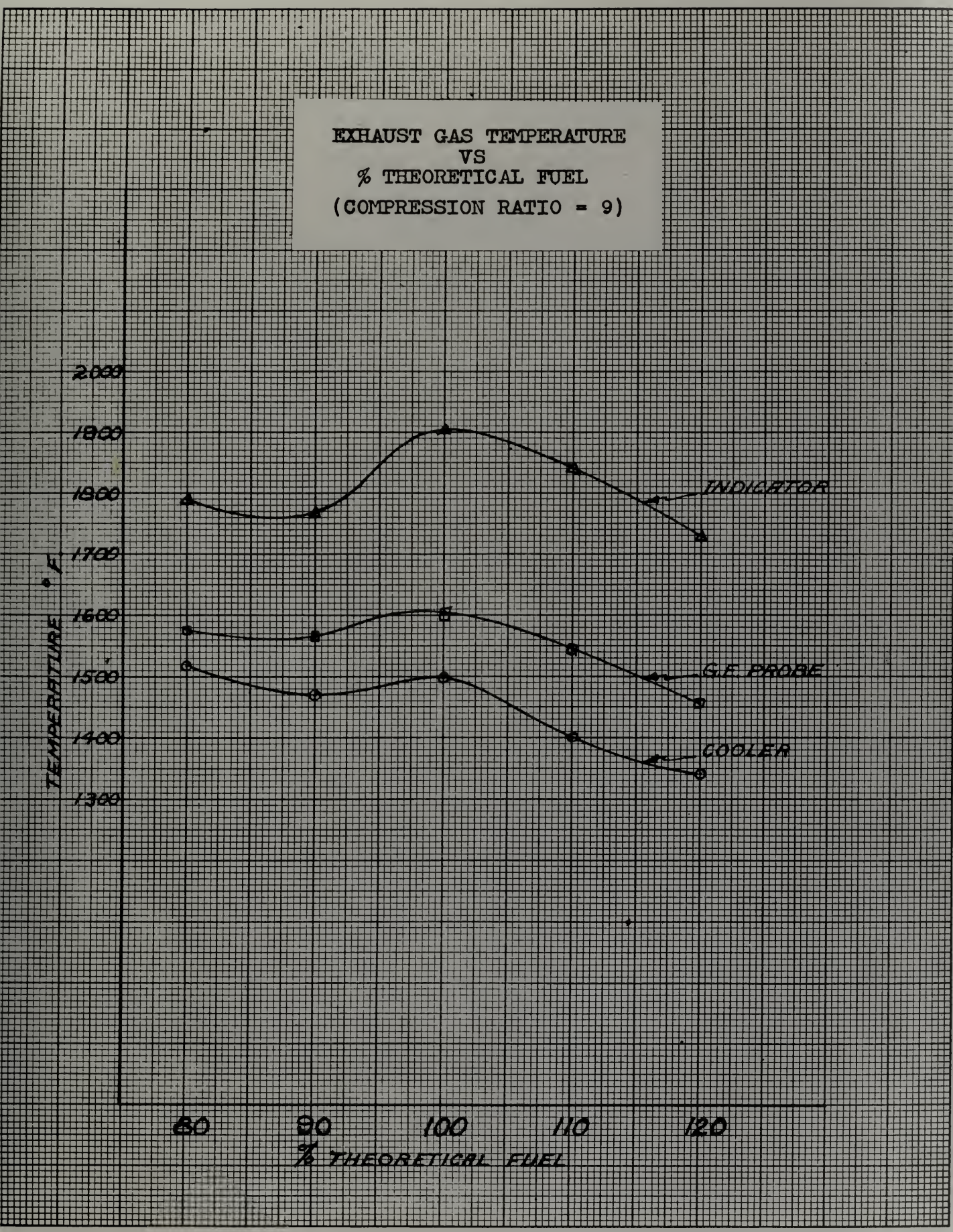

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COMPRESSION RATIO = 9)
2000
1900
1800
1700
1600
1500
1400
1300
TEMPERATURE °F
INDICATOR
G.E. PROBE
COOLER
80
90
100
110
120
% THEORETICAL FUEL

FIG. 14

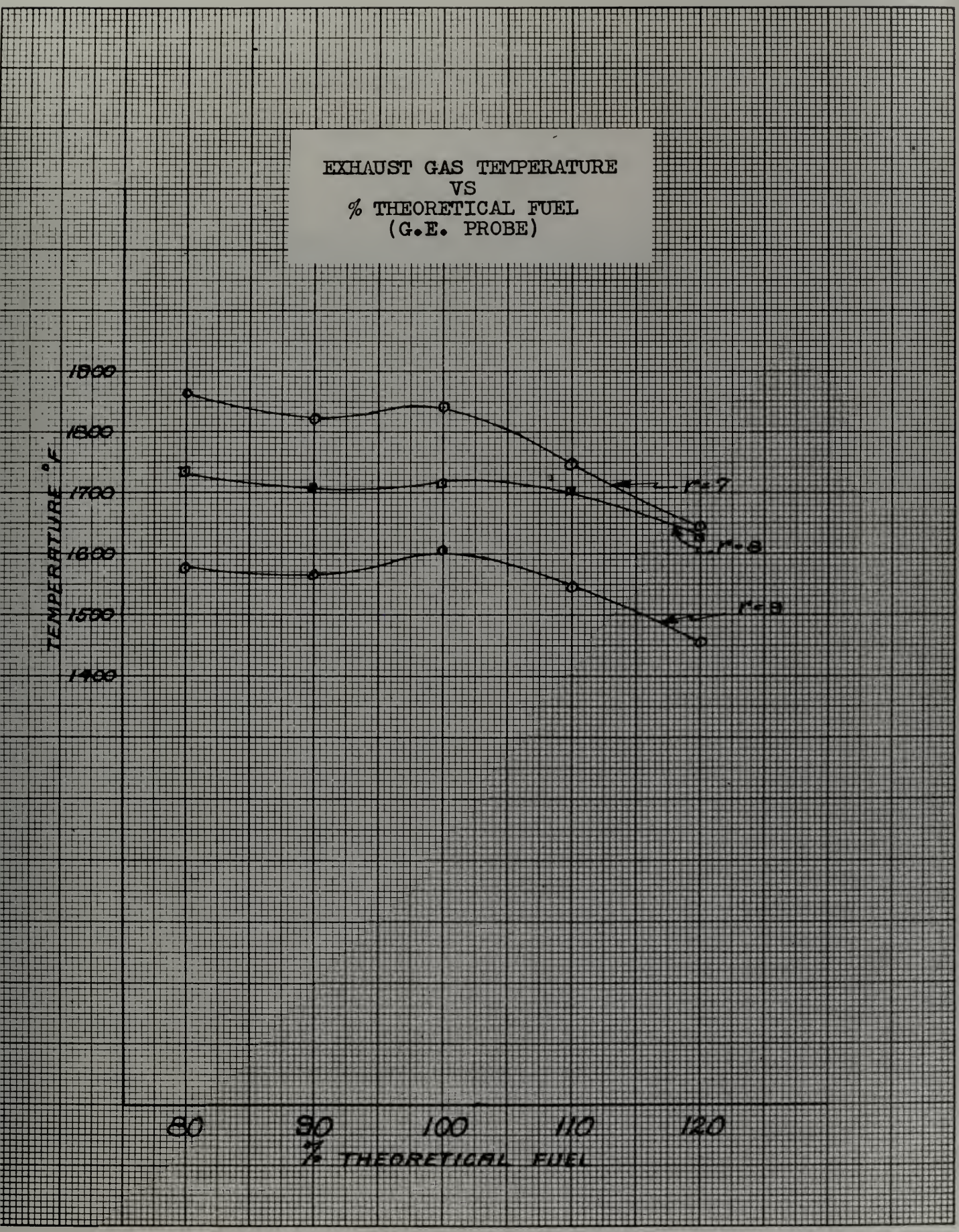

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(G.E. PROBE)
TEMPERATURE °F
1900
1800
1700
1600
1500
1400
r=7
r=8
r=9
80
90
100
110
120
% THEORETICAL FUEL

FIG. 15

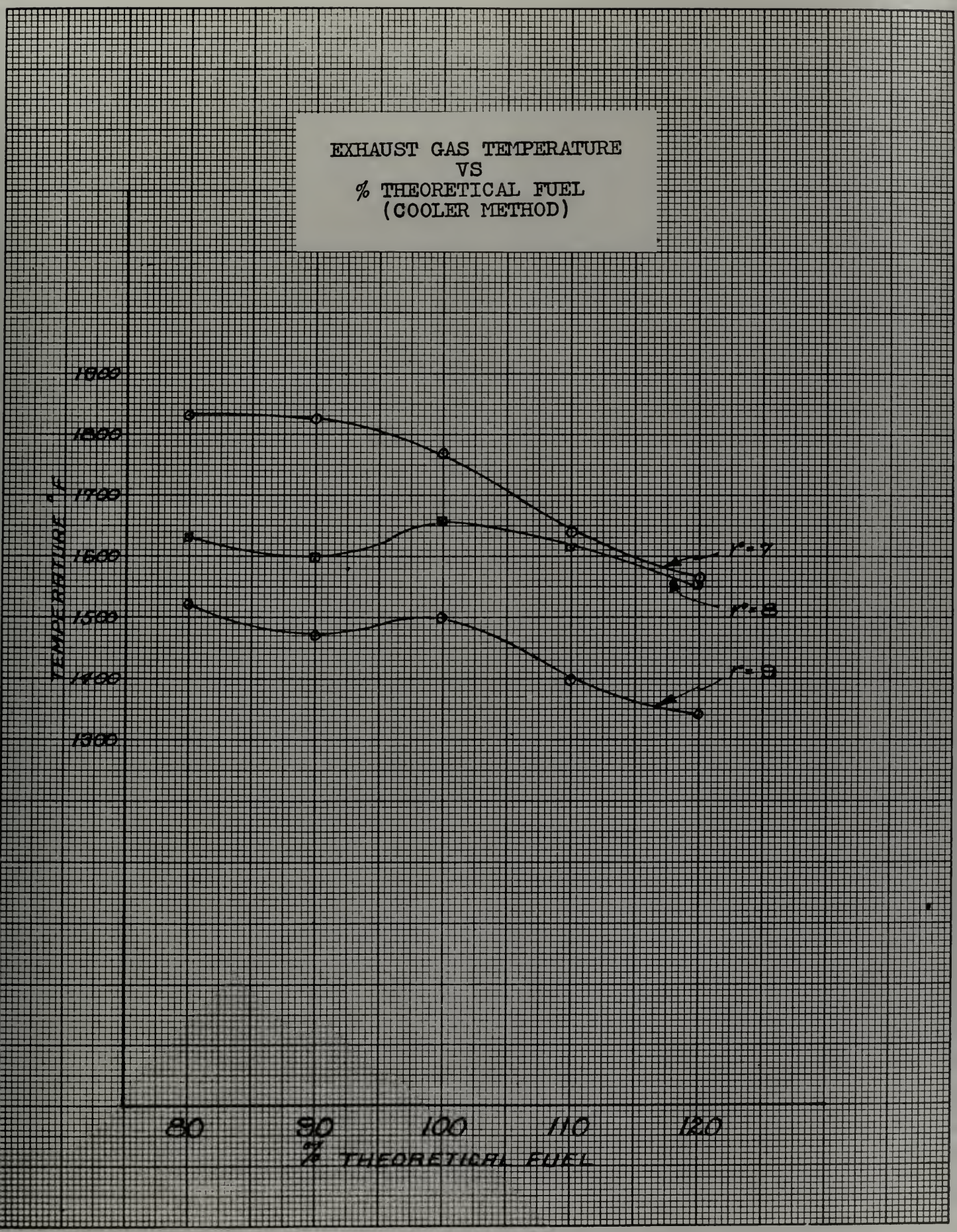

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(COOLER METHOD)
TEMPERATURE °F
1800
1700
1600
1500
1400
1300
r=7
r=8
r=9
80
90
100
110
120
% THEORETICAL FUEL

FIG. 16

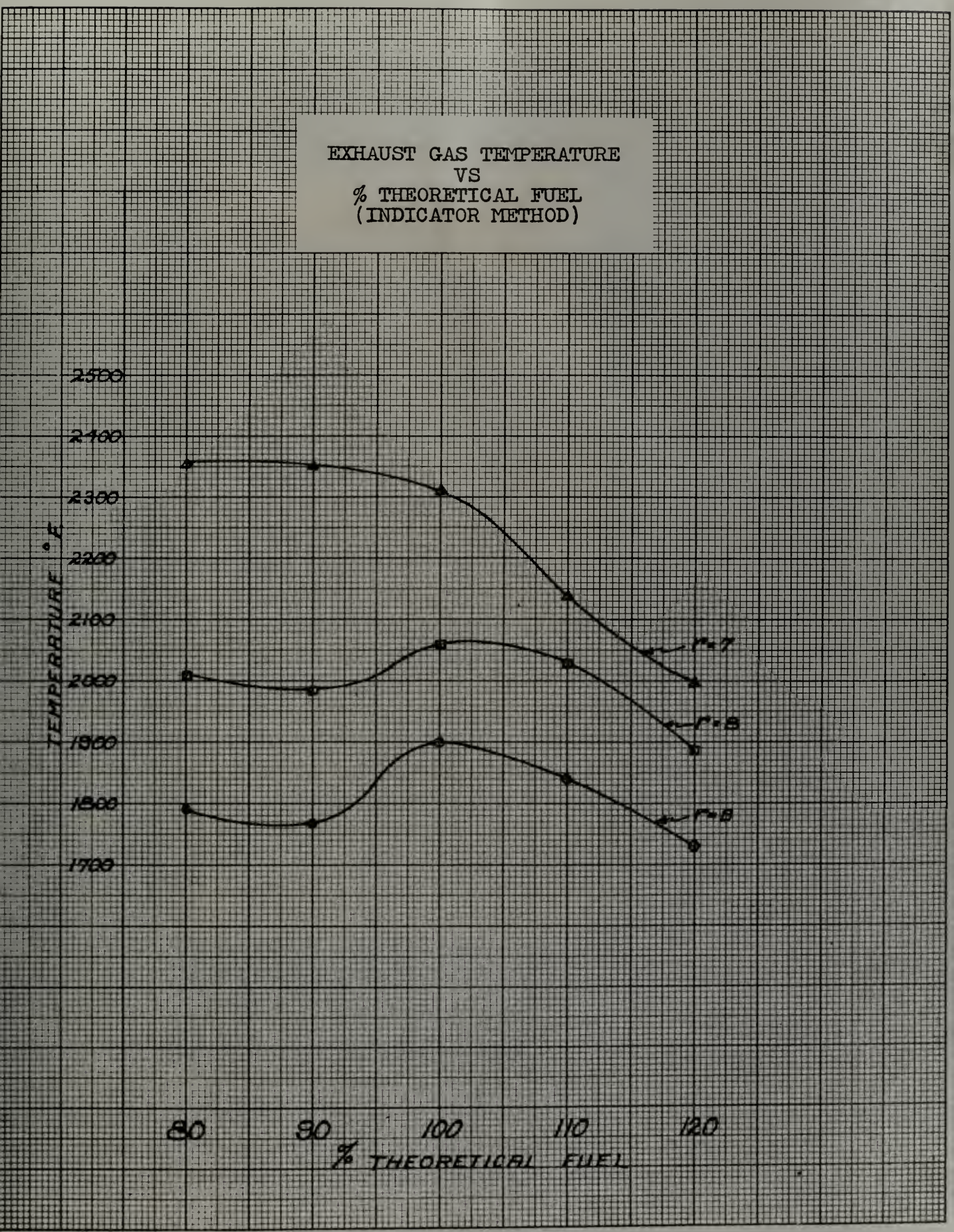

EXHAUST GAS TEMPERATURE
VS
% THEORETICAL FUEL
(INDICATOR METHOD)
TEMPERATURE °F
2500
2400
2300
2200
2100
2000
1900
1800
1700
r=7
r=8
r=9
80
90
100
110
120
% THEORETICAL FUEL

FIG. 17

DATE DUE			
27 Dec '48			

8090

Thesis
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Bennett
A study of exhaust gas temperature measurement methods.

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A study of exhaust gas temperature measu
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